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भाग 1 डिजाइन परिकलन

(पहला पुनरीक्षण)

Indian Standard

SPRINGS — DISC SPRING

PART 1 DESIGN CALCULATION

(First Revision)

ICS 21.160

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BUREAU OF INDIAN STANDARDS
MANAK BHAVAN, 9 BAHADUR SHAH ZAFAR MARG
NEW DELHI 110002

September 2004

Price Group 9

FOREWORD

This Indian Standard (Part 1) (First Revision) was adopted by the Bureau of Indian Standards, after the draft finalized by the Automotive Springs and Suspension Systems Sectional Committee had been approved by the Transport Engineering Division Council.

This standard was originally published in 1989. This revision has been undertaken due to revision of the base standard on which it was based upon.

The following technical changes have been incorporated:

a) Scope is elaborated.
b) In the design formulae for single discs, load/deflection curves for springs with or without ground ends provided.
c) Effect of friction on load/deflection characteristics provided.
d) A new clause on 'SET' is included.

Disc springs are conical-shaped annular discs which are capable of being loaded in axial direction. They may be subjected either to steady or to fatigue loading. Disc springs can be used either as single discs, or as spring assemblies consisting of discs piled on top of one another in the same direction, or as spring columns consisting of individual discs piled on top of one another in alternating directions, or as spring columns consisting of spring assemblies piled on top of one another in alternating directions. Disc springs are manufactured either with or without seating faces.

The fatigue strength values and its finite life fatigue strength values given in Fig. 12 to Fig. 14 in this standard were determined by statistical evaluation of laboratory tests conducted on test machines with uniformly sinusoidal loading. In these tests, the effect of corrosion was eliminated by adequate lubrication and by carrying out tests without any interruptions. These fatigue strength and finite life fatigue strength diagrams are plotted with a survival probability of 99 percent. This means that, out of a large number of disc springs of one range of plate thickness, irrespective of the other dimensions, the strengths specified will be attained by 99 percent of the springs without suffering a vibratory fracture. It should be mentioned in this context that the fatigue strength data given by the spring manufacturers may well deviate from the values given here, and from one another, due to the various manufacturing methods which can be used.

This part deals with design calculation of disc spring, while Part 2 covers the requirements. In the preparation of this standard, assistance has been derived from DIN 2092 : 1992 'Design of conical disc springs', issued by the Deutsches Institut für Normung (DIN).

The composition of the committee responsible for the formulation of this standard is given at Annex C.

For the purpose of deciding whether a particular requirement of this standard is complied with, the final value, observed or calculated, expressing the result of a test or analysis, shall be rounded off in accordance with IS 2 : 1960 'Rules for rounding off numerical values (*revised*)'. The number of significant places retained in the rounded off value should be the same as that of the specified value in this standard.

Indian Standard

SPRINGS — DISC SPRING

PART 1 DESIGN CALCULATION

(First Revision)

1 SCOPE

This standard (Part 1) specifies design criteria and features of conical disc springs, whether as single discs or stacks of discs. It includes the definition of relevant concepts as well as design formula, and covers the setting and endurance life of such springs.

2 REFERENCES

The following standard contains provisions which through reference in this text, constitute provision of this standard. At the time of publication, the edition indicated was valid. All standards are subject to revision and parties to agreements based on this standard are encouraged to investigate the possibility of applying the most recent edition of the standard indicated below:

IS No.	*Title*
12511 (Part 2)	Springs — Disc spring: Part 2 Specification (*first revision*)

3 CONCEPT

Disc springs are conical-shaped annular discs which are capable of being loaded in axial direction. They may be subjected either to steady or to fatigue loading. Disc springs can be used either as single discs, or as spring assemblies consisting of discs piled on top of one another in the same direction, or as spring columns consisting of individual discs piled on top of one another in alternating directions, or as spring columns consisting of spring assembles piled on top of one another in alternating directions. Disc springs are manufactured either with or without seating faces.

4 SYMBOLS

Following symbols and units shall apply (*see* Fig. 1 and Fig. 2):

D_e = Outside diameter, mm;

D_i = Inside diameter, mm;

D_o = Mean coil diameter, mm;

F_{gesR} = Spring load of the springs stacked in parallel, allowance being made for friction, N;

E = Modulus of elasticity, N/mm^2;

F = Spring force of single disc, N;

$F_1, F_2, F_3 \ldots$ = Spring forces associated with spring travels $s_1, s_2, s_3 \ldots$, N;

F_c = Calculated spring force in the pressed flat condition, N;

F_{ges} = Spring force of the spring assembly associated with spring travel s_{ges}, N;

L_o = Length of unloaded spring column or of unloaded spring assembly, mm;

$L_1, L_2, L_3 \ldots$ = Length of loaded spring column or of unloaded spring assembly associated with spring forces F_1, F_2, F_3 ...., mm;

L_c = Calculated length of spring column or of spring assembly in the pressed flat condition, mm;

N = Number of stress cycles endured up to fracture;

R = Spring rate, N/mm;

W = Work of elastic strain, N/mm;

h_o = Operand (theoretical spring travel down to the completely flat position); $h_o = l_o - t$, mm;

h^1_o = Initial cone height of springs with ground ends (equal to free overall height, $l_o - t$), mm;

i = Number of single discs or spring assemblies piled on top of one another in alternating directions to form a column;

l_0 = Overall height of unloaded single disc, mm;

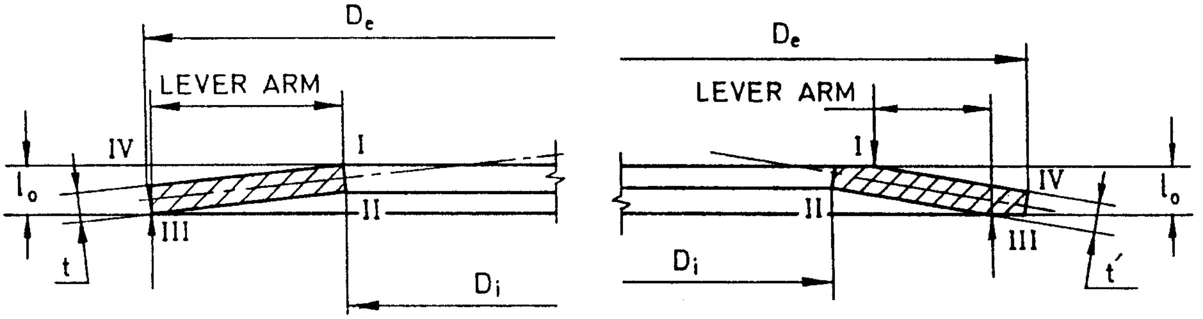
Δl = Creep, mm;

n = Number of single discs stacked on top of one another in the same direction to form an assembly;

s = Spring travel of single disc, mm;

$s_1, s_2, s_3 \ldots$ = Spring travels associated with spring forces F_1, F_2, F_3...., mm;

s_{ges} = Spring travel of the spring column or of the spring assembly not taking friction into account, mm; Recommended maximum value : $S_{ges} \approx 0.75\ (L_o - L_c)$

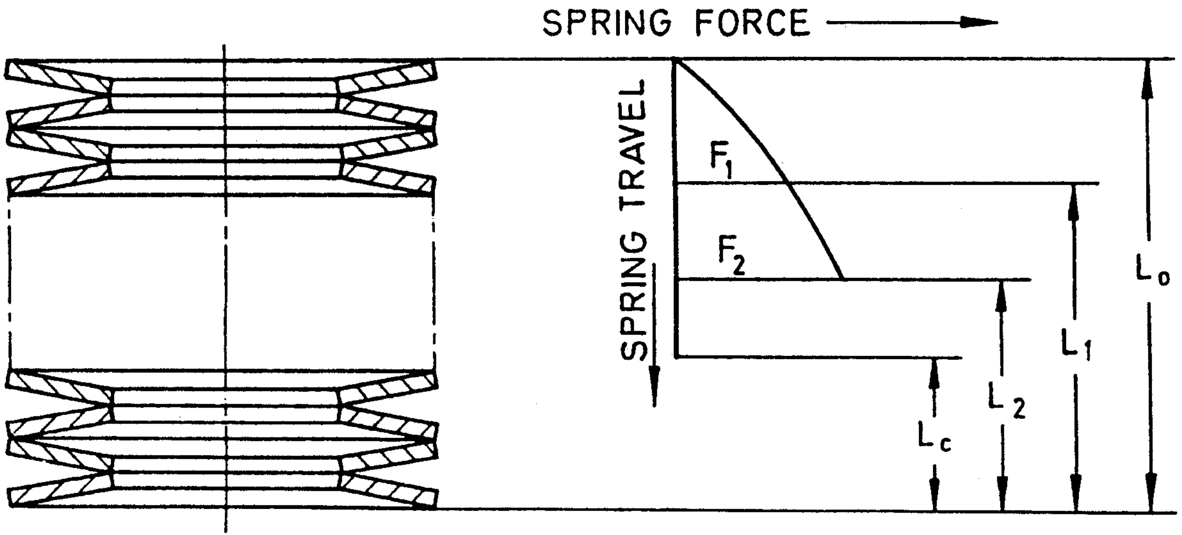


1A Without Seating Faces

1B With Seating Faces

All dimensions in millimetres.

FIG. 1 SINGLE DISC SPRING AND CROSS SECTION LOCATIONS OF THEORETICAL STRESSES



All dimensions in millimetres.

FIG. 2 EXAMPLE OF SPRING COLUMN

t = Thickness of single disc, mm;
t' = Reduced thickness of single disc in the case of disc springs with seating faces, mm;
W_M, W_R = Coefficients of friction;
σ = Theoretical stress, N/mm^2;
σ_{OM}, σ_I, σ_{II}, σ_{III}, σ_{IV} = Theoretical stresses at location OM, I, II, III, IV, N/mm^2;
σ_0 = Theoretical maximum stress of disc spring subjected to fatigue loading, N/mm^2;
σ_U = Theoretical minimum stress of disc springs subjected to fatigue loading, N/mm^2;
σ_H = Stroke stress correlated to the work travel of disc springs subjected to the fatigue loading, N/mm^2;
σ_0 = Maximum stress of fatigue limit, N/mm^2;
σ_U = Minimum stress of fatigue limit, N/mm^2;
σ_H = $\sigma_O - \sigma_U$ = fatigue stroke strength, N/mm^2;
$\left.\begin{matrix} K_1 \\ K_2 \\ K_3 \\ K_4 \end{matrix}\right\}$ = Coefficients (*see* 5);
σ = $\dfrac{D_e}{D_i}$ = Diameter ratio;
μ = Poisson's ratio, $\mu \approx 0.3$ for spring steel; and
ΔF = Relaxation.

5 FORMULAE AND COEFFICIENTS FOR THE SINGLE DISC

5.1 When calculating for the single disc spring, it must be taken into account that the effective lever (moment) arms will be dependent on the mode of force introduction during the loading of the disc spring. As a general rule, the force is introduced via location I and III as illustrated in Fig. 1A (*see* also **5.2**). However, the force can also be introduced via shortened lever arms, for example, as illustrated in Fig. 1B (*see* also **5.3**).

5.2 Single Disc Spring with Conventional Force Introduction

5.2.1 If the force is introduced via location I and III, as illustrated in Fig. 1A, the calculation shall be made as follows.

5.2.1.1 *Spring force*

$$F = \frac{4\,E}{1-\mu^2} \times \frac{t^4}{K_1 \times D_e^2} \times \frac{s}{t} \left[\frac{h_o}{t} - \frac{s}{t} \left(\frac{h_o}{t} - \frac{s}{2\,t} \right) + 1 \right] \quad \text{...(1)}$$

5.2.1.2 *Theoretical streses*

$$\sigma_{I} = \frac{4\,E}{1-\mu^2} \times \frac{t^2}{K_1 \times D_e^2} \times \frac{s}{t} \left[-K_2 \left(\frac{h_o}{t} - \frac{s}{2\,t} \right) - K_3 \right] \quad \text{...(2)}$$

$$\sigma_{II} = \frac{4\,E}{1-\mu^2} \times \frac{t^2}{K_1 \times D_e^2} \times \frac{s}{t} \left[-K_2 \left(\frac{h_o}{t} - \frac{s}{2\,t} \right) + K_3 \right] \quad \text{...(3)}$$

$$\sigma_{III} = \frac{4\,E}{1-\mu^2} \times \frac{t^2}{K_1 \times D_e^2} \times \frac{s}{t} \times \frac{1}{\delta}$$

$$\left[(2\,K_3 - K_2) \left(\frac{h_o}{t} - \frac{s}{2\,t} \right) + K_3 \right] \quad \text{...(4)}$$

$$\sigma_{IV} = \frac{4\,E}{1-\mu^2} \times \frac{t^2}{K_1 \times D_e^2} \times \frac{s}{t} \times \frac{1}{\delta}$$

$$\left[(2\,K_3 - K_2) \left(\frac{h_o}{t} - \frac{s}{2\,t} \right) - K_3 \right] \quad \text{...(5)}$$

NOTES

1 Positive values are tensile stresses and negative values are compression stresses. The stress σ_{IV} is of secondary importance only.

2 For coefficients K_1, K_2 and K_3, *see* **5.4**.

3 The value of $\frac{4\,E}{1-\mu^2}$ = 905 495 N/mm² for high grade steel with E = 206 000 N/mm².

5.2.1.3 *Spring rate*

$$R = \frac{dF}{ds} = \frac{4\,E}{1-\mu^2} \times \frac{t^3}{K_1 \times D_e^2}$$

$$\left[\left(\frac{h_o^2}{t} \right) - 3 \frac{h_o}{t} \times \frac{s}{t} + \frac{3}{2} \left(\frac{s^2}{t} \right) + 1 \right] \quad \text{...(6)}$$

5.2.1.4 *Work of elastic strain*

$$W = \int_0^s F.ds = \frac{2\,E}{1-\mu^2} \times \frac{t^5}{K_1 \times D_e^2} \left(\frac{s^2}{t} \right)$$

$$\left[\left(\frac{h_o}{t} - \frac{s^2}{2t} \right) + 1 \right] \quad \text{...(7)}$$

5.3 Single Disc Spring with Force Introduction Via Shortened Lever Arms

The force is not introduced via location I and III illustrated in Fig. 1A, but via shortened lever arms and the equations featured in **5.2** no longer apply. In the case of disc springs with seating faces, for example, Group 3 disc springs conforming to IS 12511 (Part 2), the force is also introduced via shortened lever arms (*see* Fig. 1B). On these disc springs, the disc thickness t is reduced by the manufacturer to the thickness t', in order to attain the prescribed spring force F at $s \approx 0.75\, h_o$. It can be obtained from the following formula:

$$t' = \in t$$

The factor '$\in$' depends on $\frac{h_o}{t}$ ratio (*see* Fig. 3).

5.4 Coefficients K_1, K_2 and K_3 (*see* Fig. 4 and Fig. 5 along with the table under Fig. 4)

6 CHARACTERISTIC OF A SINGLE DISC SPRING

6.1 The calculated characteristic (force/travel curve) of the single disc spring is not linear (*see* Fig. 6). Its shape is a function of the ratio h_o/t. The actual characteristic deviates only slightly from the calculated characteristic in lower portion of the spring travel range (*see* Fig. 7).

6.1.1 For $s = h_0$ the calculated spring force is:

$$F = F_c \frac{4\,E}{1-\mu^2} \times \frac{t^3 \times h_o}{K_1 \times D_e^2} \quad \text{...(8)}$$

$$K_1 = \frac{1}{\pi} \cdot \frac{\left[\frac{\delta - 1}{\delta} \right]^2}{\frac{\delta + 1}{\delta - 1} - \frac{2}{l_n \delta}}$$

6.1.2 For $s/h_0 > 0.75$, the actual characteristic also deviates increasingly from the calculated characteristic, because the disc springs begin to slip relative to each other or to the base; this steadily reduces the length of the lever arm (*see* Fig. 7).

6.1.3 When determining the characteristics by precision measurement, the reference length shall be the theoretical unloaded spring length l_o, or the column length L_o, respectively.

7 COMBINATION OF SINGLE DISC SPRINGS

7.1 Frictional Force

7.1.1 The frictional force on disc springs shall be taken

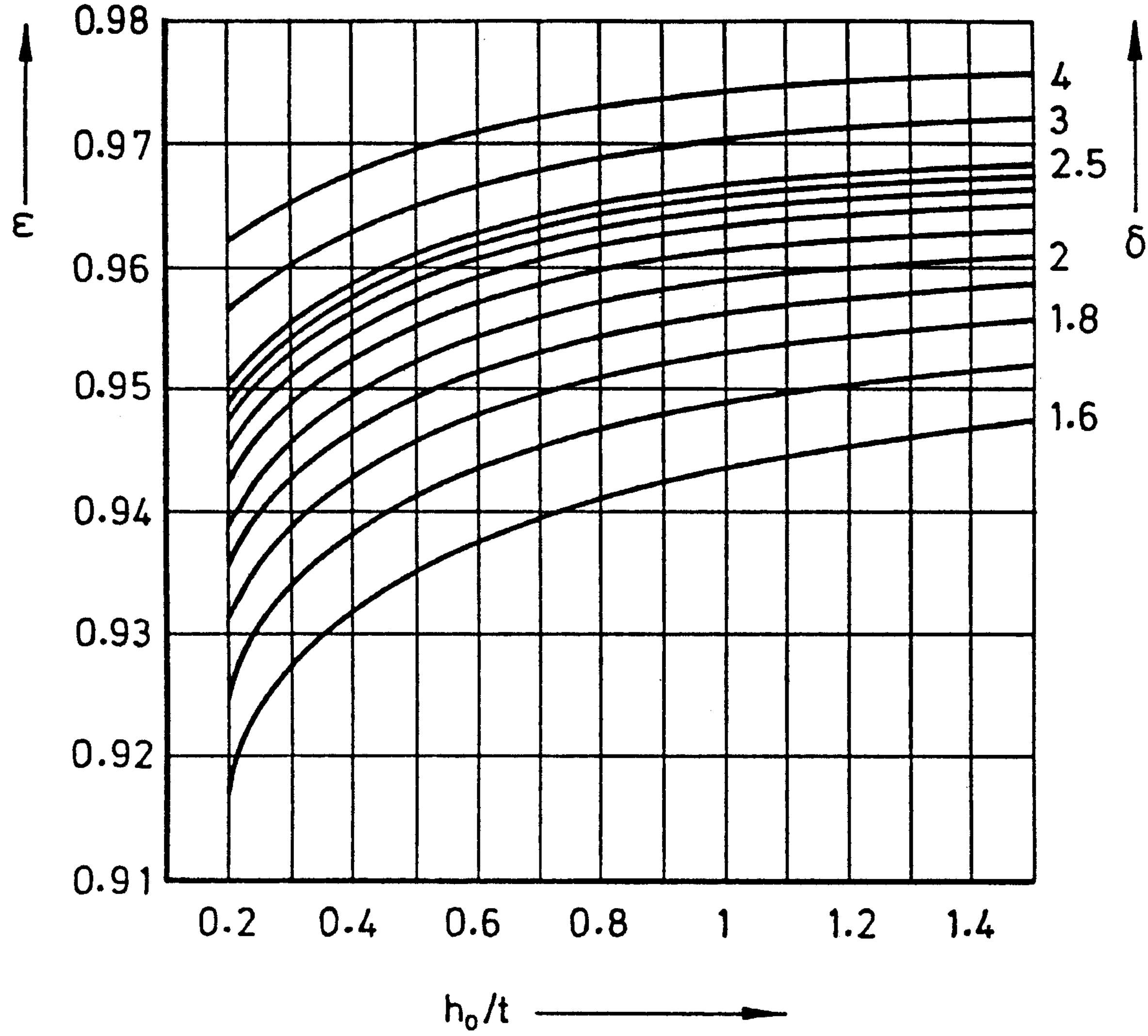


FIG. 3 VALUE OF ∈ FOR VARIOUS h_o/t RATIO

into account. It will be a function of the number of the single disc per spring assembly and of the number of springs or spring assemblies in spring column. In addition, the surface finish and lubrication may affect the frictional force. Due to the friction, the load values are likely to vary and the average variation expected is as under:

a) Single disc assembly : ± 2% to 3%
b) Double disc assembly : ± 4% to 6%
c) Triple disc assembly : ± 6% to 9%
d) Quadruple disc assembly : ± 8% to 12%

7.2 Spring Assembly

7.2.1 The spring assembly consists of single disc springs stacked on top of one another in the same direction as shown in Fig. 8.

7.2.1.1 In case of n single discs stacked on top of one another in the same direction, ignoring friction, the following applies:

$$F_{ges} = n \times F \quad \text{...(9)}$$

$$s_{ges} = s \quad \text{...(10)}$$

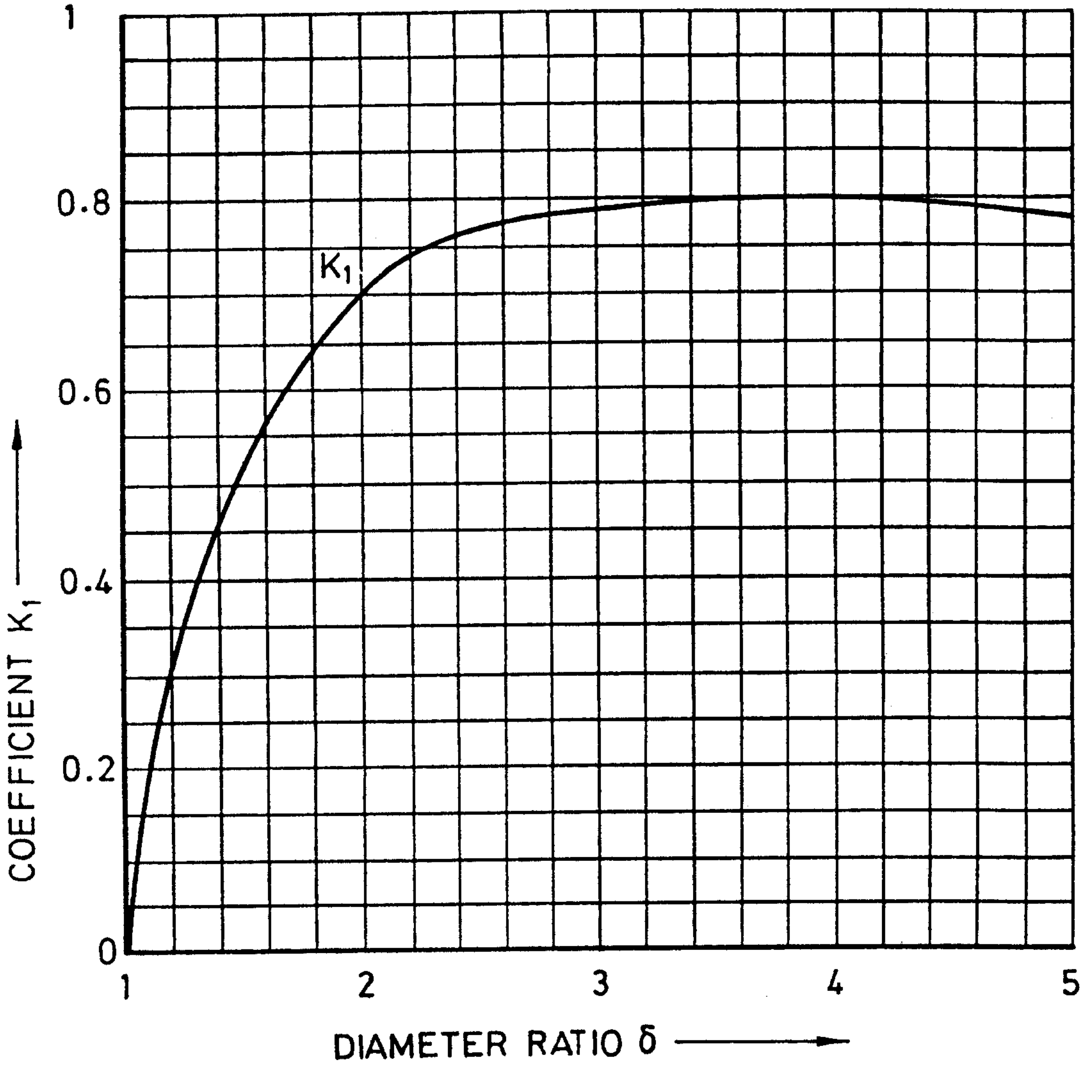
$$L_o = l_o + (n-1)\,t \quad \text{...(11)}$$

7.3 Spring Column

The spring column consists of single disc springs or spring assemblies piled on top of one another in alternating directions.

7.3.1 Spring column consisting of single discs is shown in Fig. 9.

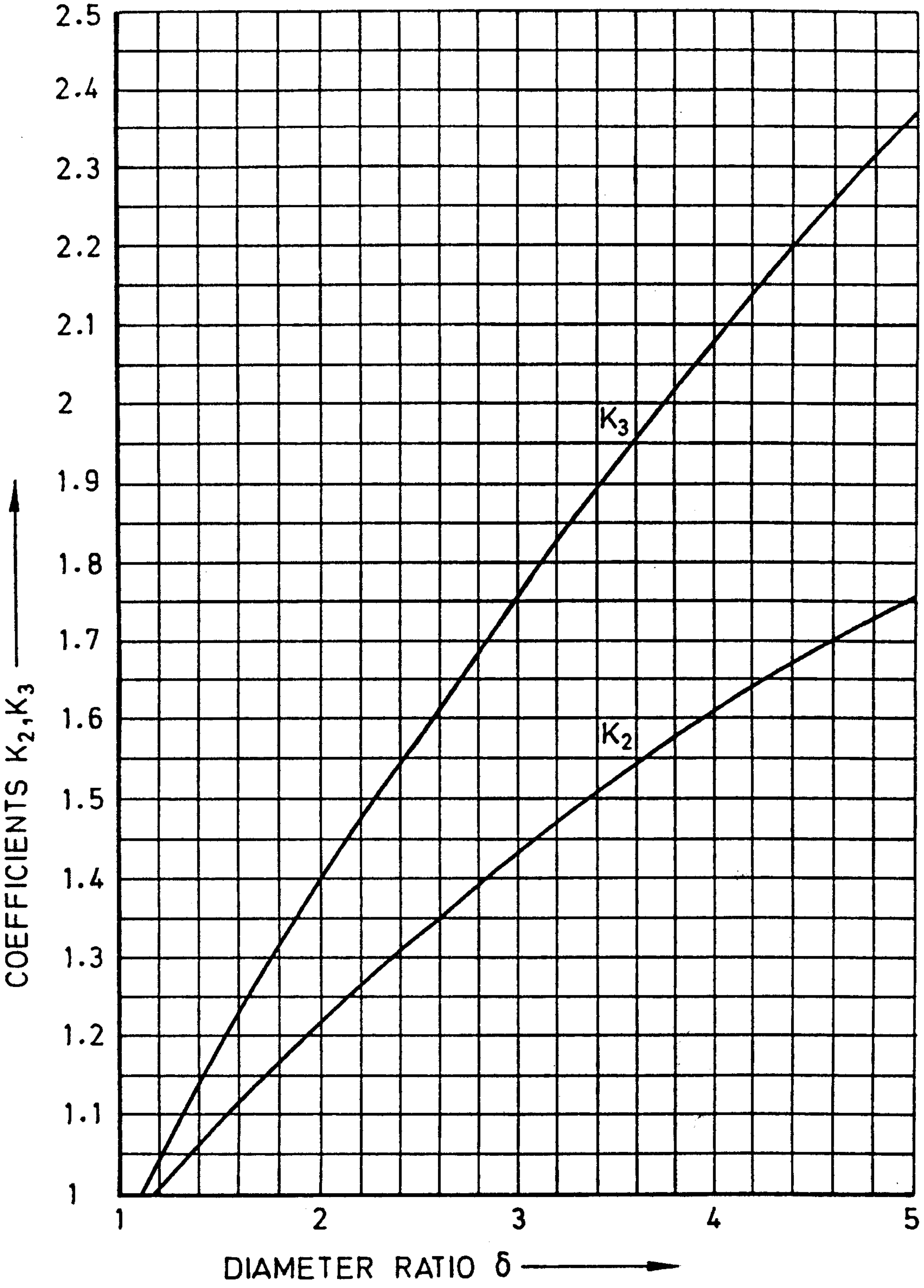
7.3.1.1 In case of i single disc springs piled on top of one another in alternating directions, ignoring friction, the following applies:



$$K_1 = \frac{1}{\pi} \frac{\left(\frac{\delta-1}{\delta}\right)^2}{\frac{\delta+1}{\delta-1} - \frac{2}{l_n \cdot \delta}}$$

δ	K_1	K_2	K_3	δ	K_1	K_2	K_3
1.2	0.29	1.02	1.05	2.7	0.77	1.37	1.63
1.3	0.39	1.04	1.09	2.8	0.78	1.39	1.67
1.4	0.46	1.07	1.14	2.9		1.41	1.70
1.5	0.52	1.10	1.18	3.0		1.43	1.74
1.6	0.57	1.12	1.22	3.1	0.79	1.45	1.77
1.7	0.61	1.15	1.26	3.2		1.46	1.81
1.8	0.65	1.17	1.30	3.4		1.50	1.87
1.9	0.67	1.20	1.34	3.6	0.80	1.54	1.94
2.0	0.69	1.22	1.38	3.8		1.57	2.00
2.1	0.71	1.24	1.42	4.0		1.60	2.07
2.2	0.73	1.26	1.45	4.2	0.80	1.64	2.13
2.3	0.74	1.29	1.49	4.4		1.67	2.19
2.4	0.75	1.31	1.53	4.6	0.80	1.70	2.25
2.5	0.76	1.33	1.56	4.8	0.79	1.73	2.31
2.6	0.77	1.35	1.60	5.0		1.76	2.37

FIG. 4 COEFFICIENT K_1 AS A FUNCTION OF DIAMETER RATIO δ



$K_2 = \frac{1}{\pi} \times \frac{6}{l_n \cdot \delta} \times \left(\frac{\delta - 1}{l_n \cdot \delta} - 1 \right)$

$K_3 = \frac{1}{\pi} \times \frac{6}{l_n \cdot \delta} \times \frac{\delta - 1}{2}$

FIG. 5 COEFFICIENTS K_2 AND K_3 AS A FUNCTION OF DIAMETER RATIO δ

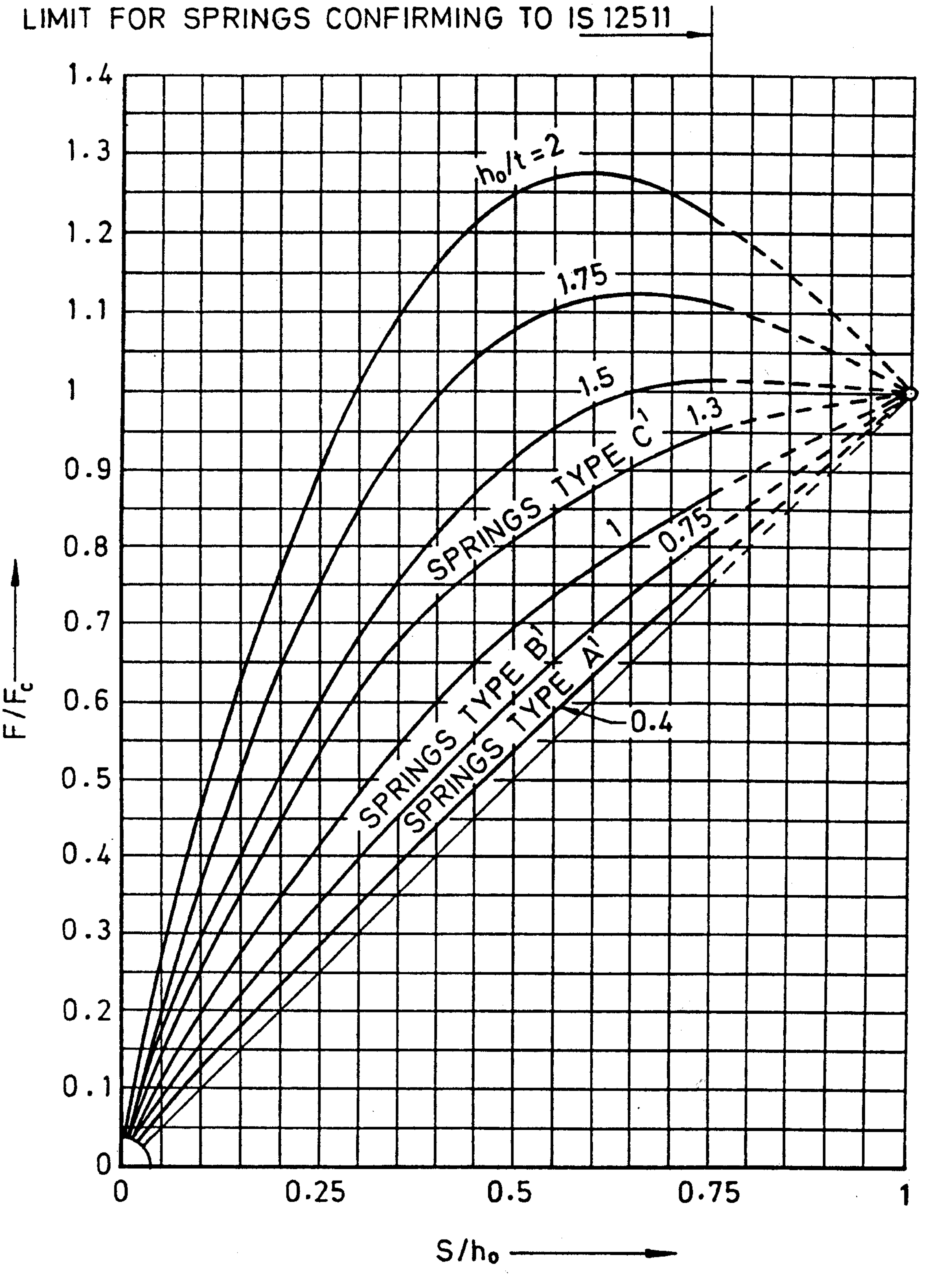


FIG. 6 BEHAVIOUR OF CALCULATED SPRING CHARACTERISTIC FOR VARIOUS h_o/t RATIO

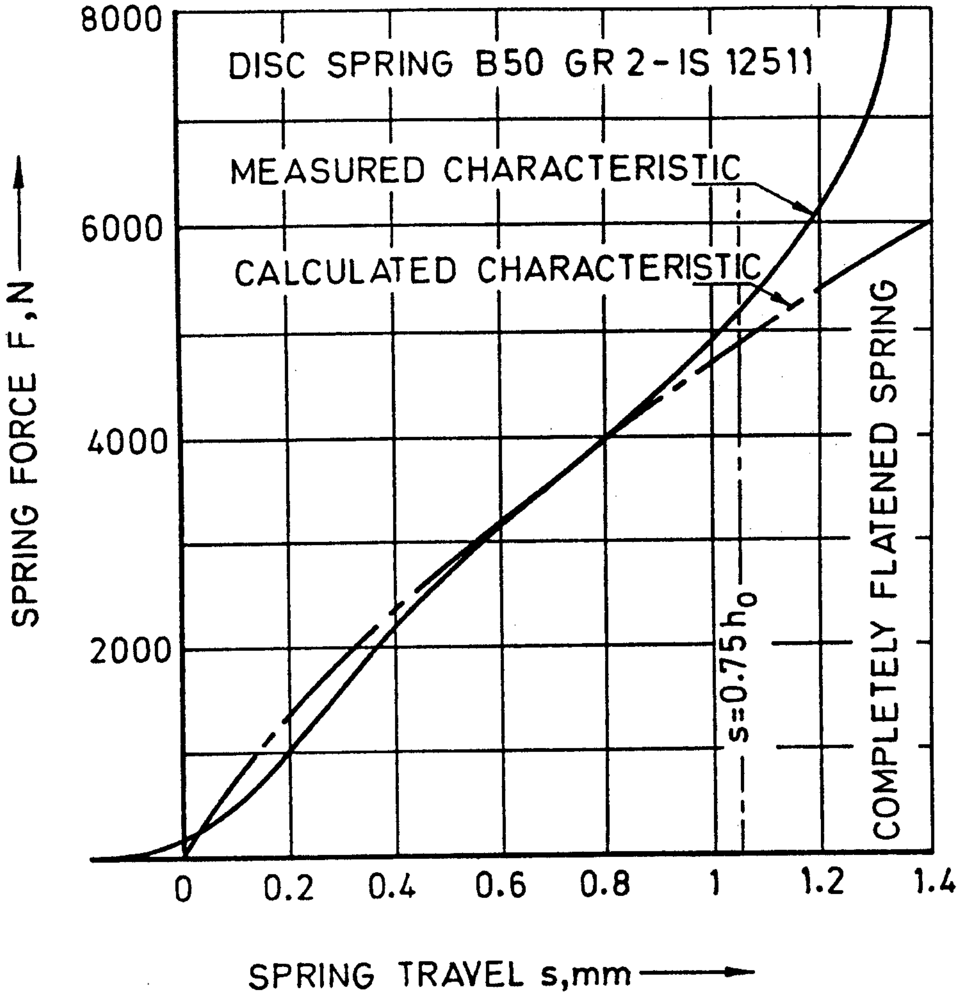


FIG. 7 CALCULATED AND MEASURED CHARACTERISTIC OF A DISC SPRING

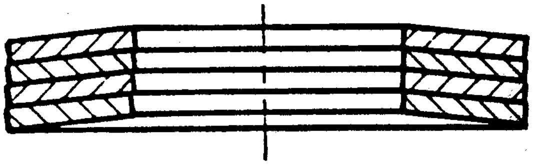


FIG. 8 SPRING ASSEMBLY (SINGLE DISC SELECTED ON TOP OF ONE ANOTHER IN SAME DIRECTION)

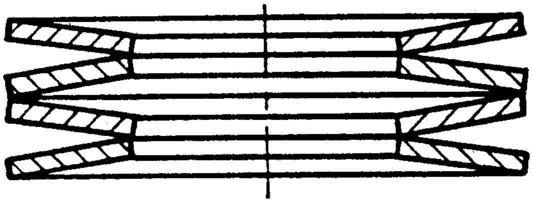


FIG. 9 SPRING COLUMN (SINGLE DISC PILED ON TOP OF ONE ANOTHER IN ALTERNATING DIRECTION)

$$F_{ges} = F \qquad ...(12)$$

$$s_{ges} = i \times s \qquad ...(13)$$

$$L_0 = i \times l_0 \qquad ...(14)$$

7.3.2 Spring column consisting of spring assemblies is shown in Fig. 10.

7.3.2.1 In case of *i* spring assemblies piled on top of one another in alternating directions, each assembly consisting of *n* single springs, ignoring friction, the following applies:

$$F_{ges} = n \times F \qquad ...(15)$$

$$s_{ges} = i \times s \qquad ...(16)$$

$$L_0 = i\,[l_0 + (n-1)\,t] \qquad ...(17)$$

7.4 Different Arrangements of Single Disc Springs

7.4.1 Disc spring can be combined to form spring column in many different ways.

7.4.2 In the case of disc springs with a ratio h_0/t of more than 1.3 approximately, it is likely that the degree of flattening of individual disc springs in the spring column will vary from spring to spring.

7.4.3 In case of single disc springs stacked on top of one another in the same direction, the spring force for an equal spring travel is directly proportional to the number of single disc springs in the assembly (*see* Fig. 11A and Fig. 11B).

7.4.4 In the case of single disc springs piled on top of one another in alternating directions, the spring travel for an equal spring force is directly proportional to the number of single disc springs (*see* Fig. 11A and Fig. 11C).

7.4.5 In the case of spring assemblies piled on top of one another in alternating directions (*see* Fig. 11D), the spring force will increase with the number of individual disc springs per spring assembly and the spring travel will increase with the number of individual disc springs per spring assembly, and the spring travel will increase with the number of spring assemblies.

7.4.6 When single disc springs of different thicknesses are combined to form a spring column (*see* Fig. 11E), a characteristic curve with a steep slope can be obtained. The same effect can be achieved by using single disc springs, all of the same thickness combined into spring assembly with an increasing number of single disc springs per assembly and piled on top of one another to form a spring column (*see* Fig. 11F). However, in case of these last two combinations, the maximum permissible stress of the springs must be taken into account for portion 1 and 2 of the spring column, and the clear height h_0 in the disc spring must be altered or any exceeding of the spring travel $s \approx 0.75\ h_0$ must be avoided by appropriate measure (spacer rings, stepped guide pins).

NOTE — The frictional force has not been taken into account in Fig. 11A to Fig. 11F.

8 EFFECT OF FRICTION ON LOAD/DEFLECTION CHARACTERISTICS

When designing springs, friction shall be accounted for. The associated load component is a function of the number of single discs or elements making up a stack of springs. In this regard, surface finish and lubrication are also of relevance.

8.1 Springs Stacked in Parallel

8.1.1 In the case of springs stacked in parallel, frictional forces act between the conical contact surfaces of the individual elements (factor W_M), and between the points of loading designated I and III and the contact surfaces of the flat plates between which the spring is compressed (factor W_R). Such forces have the effect of increasing the spring load when the spring is loaded and decreasing it when the load is removed. The load/deflection characteristic shall be calculated using the following equation:

$$F_{gesR} = F \times \frac{n}{1 \pm W_m\,(n-1) \pm W_R} \qquad ...(18)$$

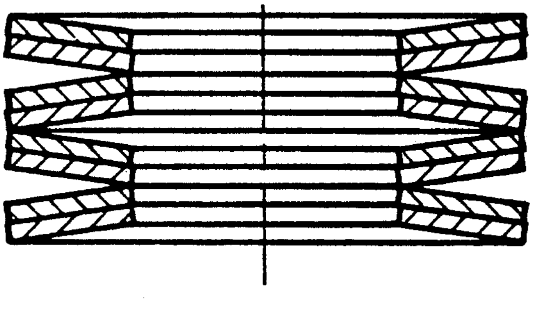


FIG. 10 SPRING COLUMN CONSISTING OF SPRING ASSEMBLIES (SPRING ASSEMBLIES PILED ON TOP OF ONE ANOTHER IN ALTERNATING DIRECTIONS TO FORM A SPRING COLUMN)

where

F = spring load determined in accordance with equation 1;

n = number of elements making up the stack of springs;

W_M = factor to account for inter-surface friction (*see* Table 1);

W_R = factor to account for edge friction (*see* Table 1);

– = indicates the loaded state; and

+ = indicates the unloaded state.

Table 1 Values of Friction for Various Groups of Springs

(*Clause* 8.1.1)

Sl No.	Spring Series	W_M	W_R
1)	(2)	(3)	(4)
i)	A	0.005 - 0.03	0.03 - 0.05
ii)	B	0.003 - 0.02	0.02 - 0.04
iii)	C	0.002 - 0.015	0.01 - 0.03

The equation (18) also accounts for the frictional behaviour of a single disc.

8.1.2 Within permissible tolerances, the actual spring will deviate from the geometrically ideal form. In the case of springs stacked in parallel, such inevitable deviation results in an actual load/deflection curve that is different from the theoretical curve, particularly in the lower range of the curve.

8.2 Springs Stacked in Series

8.2.1 In the case of multiples of springs stacked in series, the non-uniform load distribution between the single discs making up each element results in relative transverse displacement,which again results in bearing forces along the mandrel that are too high. Such forces result in frictional loss, particularly at the ends of the spring when it is deflecting. The satisfactory method of dealing with the actual load/deflection curves obtained, which deviate significantly from the theoretical curves, has not yet been devised. At present, it is only possible to accurately account for inter-surface friction, as follows:

$$F_{gesR} = F \times \frac{n}{1 \pm W_m\ (n-1)} \qquad ...(19)$$

and

$$S_{ges} = i.s \qquad ...(20)$$

where

F = spring load determined in accordance with equation (1);

n = number of single discs making up each element;

i = number of elements making up the stack of springs;

W_M = factor to account for inter-surface friction (*see* Table 1);

– = indicates the loaded state; and

+ = indicates the unloaded state.

8.2.2 In the situation when little friction is desired, single discs should be stacked in series.

9 CALCULATION OF DISC SPRINGS SUBJECTED TO STEADY LOADING OR TO SELDOM ALTERNATING LOADING

9.1 Steady loading or seldom alternating loading applies:

a) In the case of disc springs which are stressed only statically without any load alteration; and

b) In the case of disc springs which are subjected only to occasional load alteration at widely spaced time intervals, and to less than 10^4 alternating load cycles during their entire planned service life.

9.2 Critical Cross-Section Location

9.2.1 In case of disc springs subjected to steady loading or to seldom alternating loading, the maximum stress which occurs shall be taken, that is the theoretical stress σ_I at the top inner rim of the single disc spring (*see* Fig. 1). This stress shall be calculated in accordance with equation (2) if the face is introduced in the conventional way (*see* **5.2**).

9.3 Permissible Stresses

9.3.1 In the case of disc springs conforming to IS 12511 (Part 2) subjected to steady loading or to seldom alternating loading, the theoretical stress σ_I at the top inner rim shall not exceed the following guideline values:

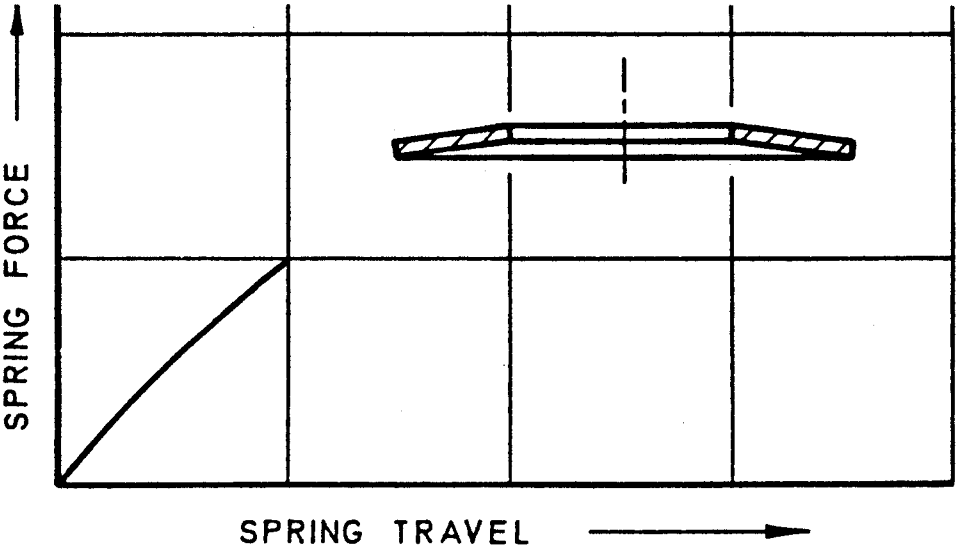
a) For $s \approx 0.75\ h_0 : \sigma_I = 2\ 000$ to $2\ 400\ N/mm^2$, and

b) For $s \approx = h_0 : \sigma_I = 2\ 600$ to $3\ 000\ N/mm^2$.

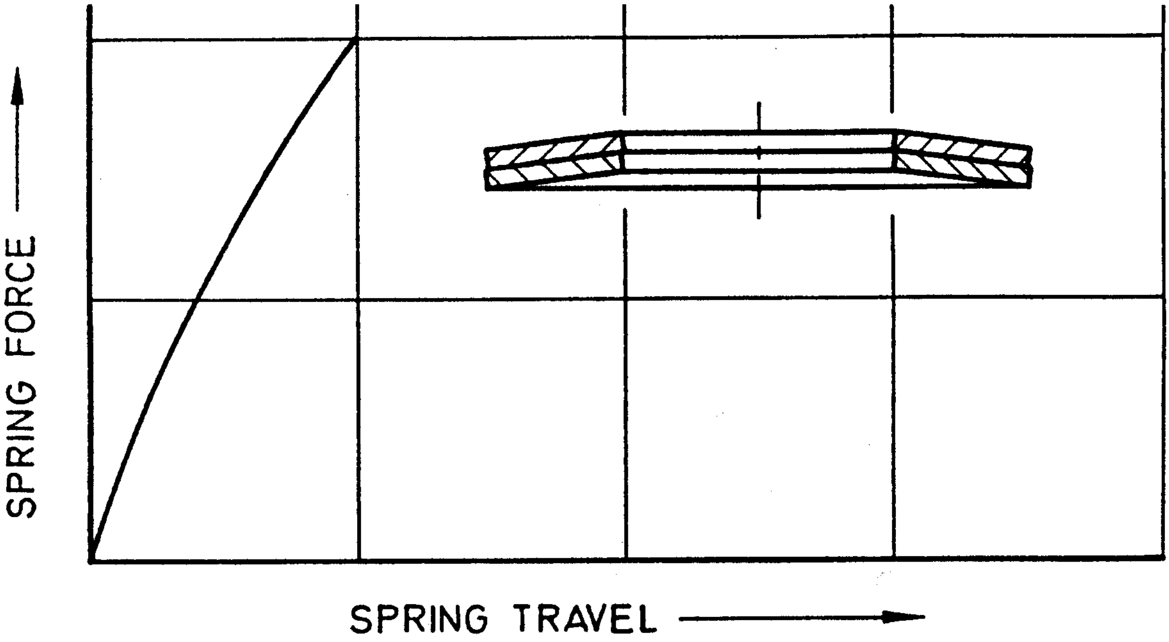
9.3.1.1 At higher stresses, the disc springs are liable to set unduly.

9.3.1.2 Disc spring conforming to IS 12511 (Part 2) can be stressed steadily or under seldom alternating load conditions up to $s \approx 0.75\ h_0$ without having to fear any setting effects; this means that as a general rule, it will not be necessary to check their theoretical stress, if they are loaded in this way.

9.3.1.3 When materials other than that listed in



11A



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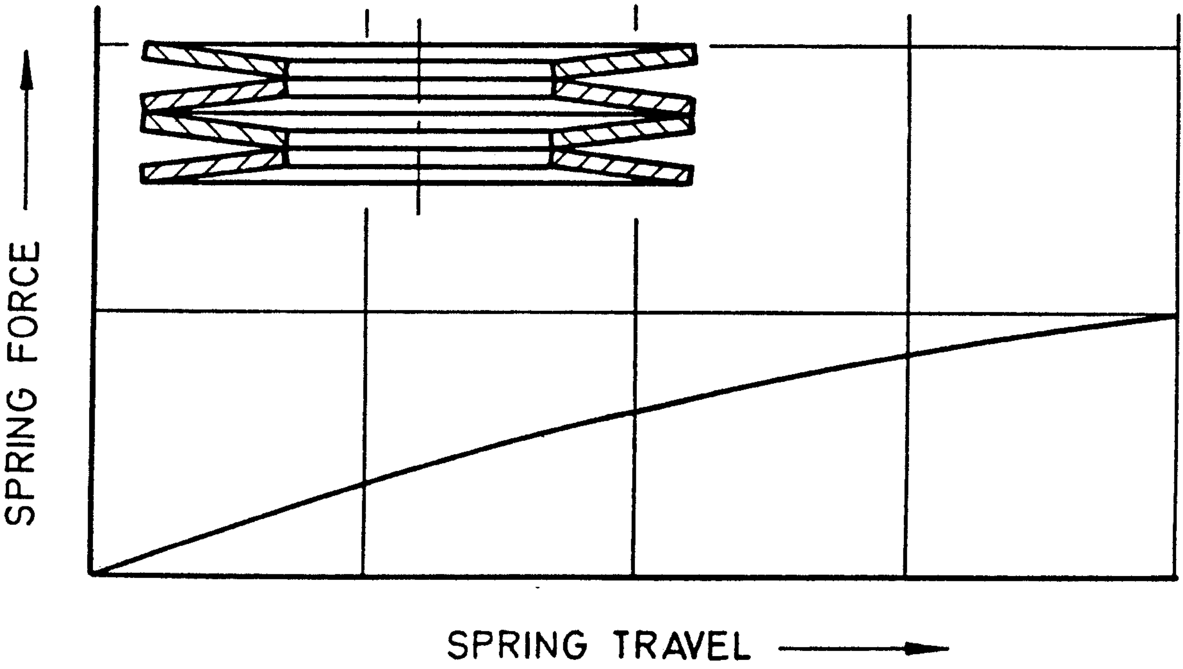
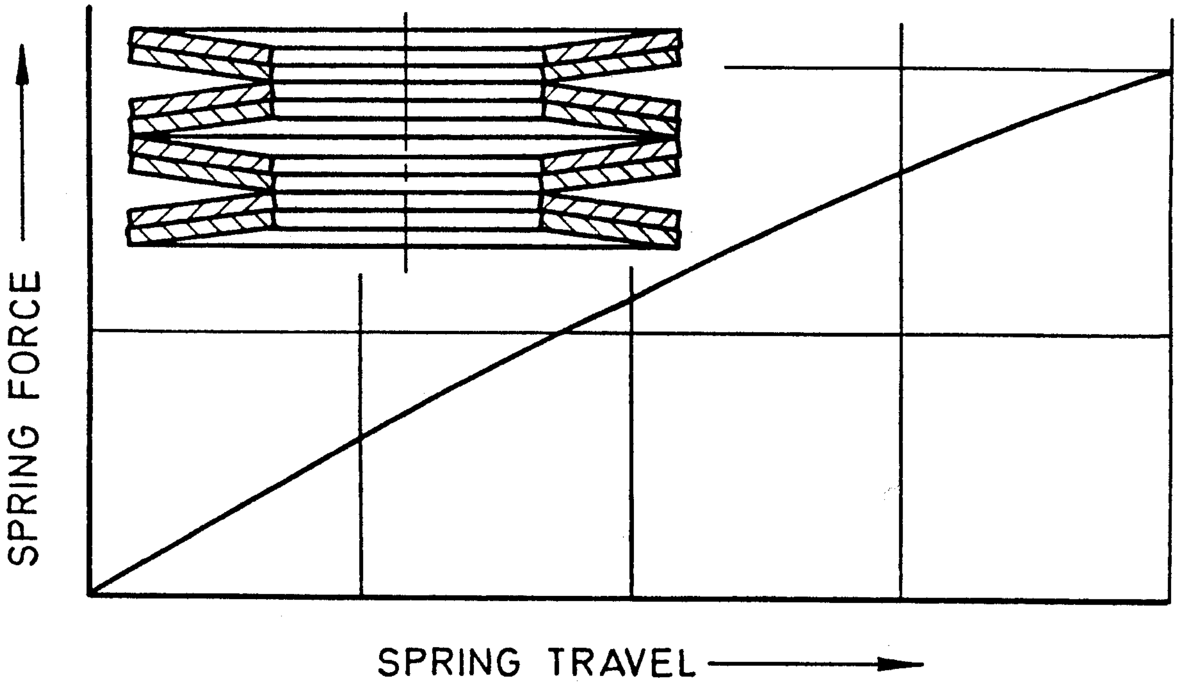
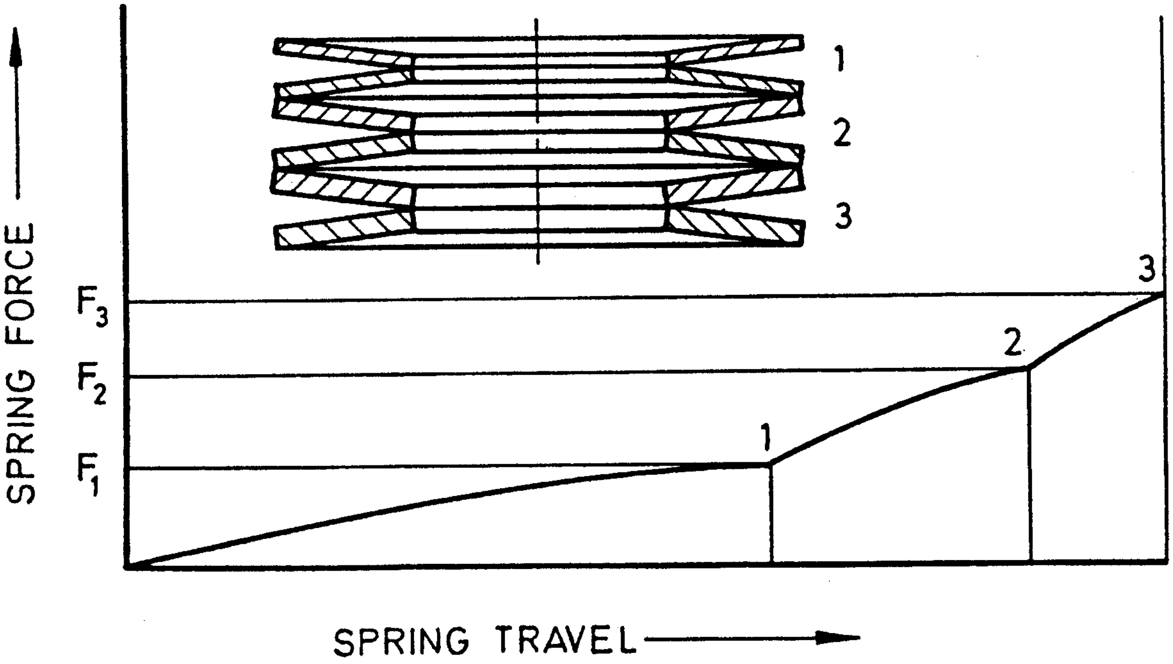


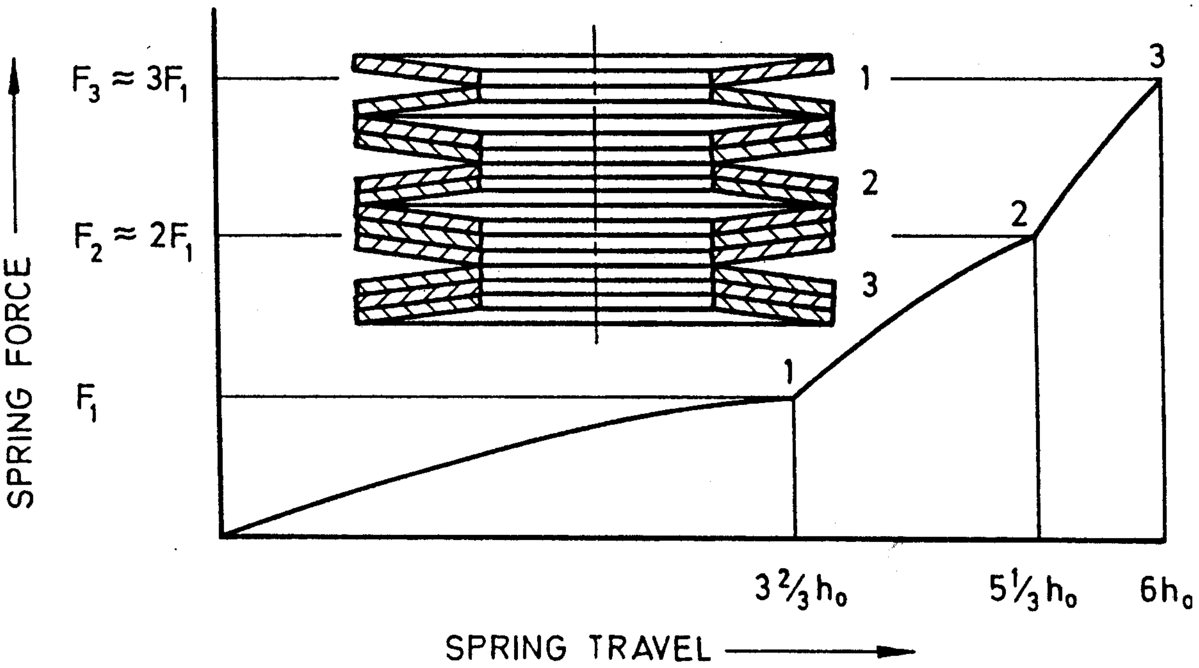
11C



11D



11E



11F

FIG. 11 CHANGE IN CALCULATED CHARACTERISTICS BY ARRANGING THE SINGLE DISC SPRINGS OF THE SPRING COLUMN IN DIFFERENT WAYS AND BY SELECTING DISC SPRINGS OF DIFFERENT THICKNESS

IS 12511 (Part 2) are used, it is advisable to consult the spring manufacturer.

9.4 Examples of calculation are given in Annex A.

10 CALCULATION OF DISC SPRING SUBJECTED TO VIBRATORY LOADING

10.1 Vibratory loading applies in all cases of disc springs in respect of which the loading alternates continuously between the prestressing spring travel s_1 and a spring travel s_2. Under the influence of the stroke stress σ_H, disc springs subjected to vibratory loading can be subdivided into groups depending on the length of their service life which in turn depends on the spring design.

a) Disc springs with a practically unlimited service life shall be capable of withstanding 2×10^6 or more alternating load cycles without fracture.

b) Disc springs with a limited service life shall be able to withstand a limited number of alternating load cycles before fracture, within the range of fatigue strength.

$$10^4 \leq N < 2 \times 10^6$$

10.2 Critical Cross-Section Location for Vibratory Fracture

10.2.1 In the case of disc springs subjected to vibratory loading, the theoretical tensile stresses at the underside of the single disc are determining. The cross-section location II or III (*see* Fig.12) with greatest theoretical stress is the critical one for vibratory fracture. Which of these two cross-section locations will in fact be the one concerned will depend to a great extent on the ratios D_e/D_i and h_0/t.

10.2.2 The stress σ_{II} and σ_{III} respectively are calculated in case of conventional force introduction (*see* **5.2**):

a) In accordance with equation (3) for cross-section location II; and

b) In accordance with equation (4) for cross-section location III.

IS 12511 (Part 2) lists the theoretical maximum tensile stresses at the underside of the standardized single disc spring for $s \approx 0.75\ h_0$; therefore an approximate evaluation of the theoretical maximum stresses for these springs at other spring travels is possible by interpolation.

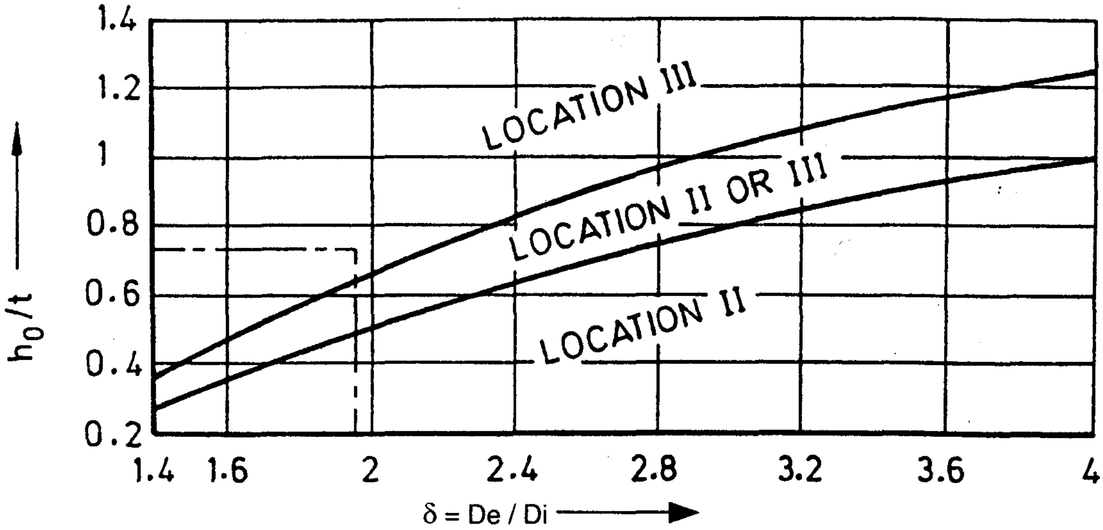
Chain-dotted lines depict the example conforming to **B-1.1**.

10.3 Minimum Prestressing Spring Travel for Avoiding Incipient Cracks

Disc springs subjected to vibratory loading shall be installed with prestressing spring travel of at least $s_1 = 0.15\ h_0$ to $0.20\ h_0$ in order to prevent the occurrence of incipient cracks at the cross-section location I (*see* Fig. 1) as a result of internal tensile stresses from the setting process.

10.4 Permissible Loadings and Permissible Range of Spring Travel

10.4.1 *Fatigue Strength Values and Finite Life Fatigue Strength Values*



Chain dotted lines depict the example conforming to **B-1.1**

FIG. 12 CRITICAL CROSS-SECTION LOCATION FOR THE VIBRATING FATIGUE IN CASE OF GROUP 1 AND GROUP 2 DISC SPRING CONFORMING TO IS 12511 (Part 2) [*see* FIG. 1A]

10.4.1.1 Figure 11C to Figure 11E were drawn on the basis of statistical evaluation of laboratory tests on testing machines with a uniform sinusoidal loading, assuming a survival probability of 99 percent. The graphs apply to single disc springs and to spring columns with $i < 6$ single disc springs piled on top of one another in alternating directions, working at standard room temperature, with a surface-hardened and impeccably machined inner or outer guide, and a minimum prestressing spring travel $s_1 = 0.15\ h_0$ to $0.20\ h_0$.

10.4.1.2 In order to avoid any unnecessary reduction in the length of service life, the disc springs must be protected from mechanical damage or other potentially harmful outside influences.

10.4.1.3 In actual practice, it must be borne in mind that the mode of stressing deviates in many cases from an approximately sinusoidal vibration. In the case of shock, such as alternating loading and as a result of natural vibration, the length of service life may be reduced.

10.4.1.4 The values of the graphs shall only be used in conjunction with appropriate safety factors in such cases of loading. If necessary, the spring manufacturer should be consulted.

NOTES

1 In the fatigue limit and finite life fatigue strength diagrams (Fig. 13 to Fig. 15), guideline values for the fatigue stroke strength σ_H for $N > 2 \times 10^6$ and for the finite life fatigue strength for $N = 10^5$ and $N = 5 \times 10^5$ as a function of the minimum stress σ_U, are specified for disc springs subjected to vibratory loading. Intermediate values for other load cycle number can be estimated.

2 There are no reliable and satisfactory fatigue strength values available today for disc springs made from material other than those specified with IS 12511 (Part 2) and for spring columns with $i > 6$, or consisting of multiple-stacked single disc springs, and also in respect of other unfavourable influences which can also be of thermal or chemical nature. On request, the spring manufacturers can provide guidance in such cases.

10.4.2 *Determination of Permissible Work Travel*

The spring travel s_2, must not exceed the corresponding value of the fatigue stroke strength σ_H or of the specified finite life fatigue strength in accordance with Fig. 13 to Fig. 15. In each case, the maximum calculated tensile stress at the underside of the individual disc spring shall be determined as a function of the spring travel (*see* **10.2** and Fig. 1). From the behaviour of the **10.4.2.1** the stroke stress σ_h which is comprised between the prestressing spring travel s_1 and stress curve plotted against the spring travel, the associated stroke stress σ_h can be read off for each work travel, and compared with the fatigue strength diagram or the finite life fatigue strength diagram.

10.5 Examples of calculation are given in Annex B.

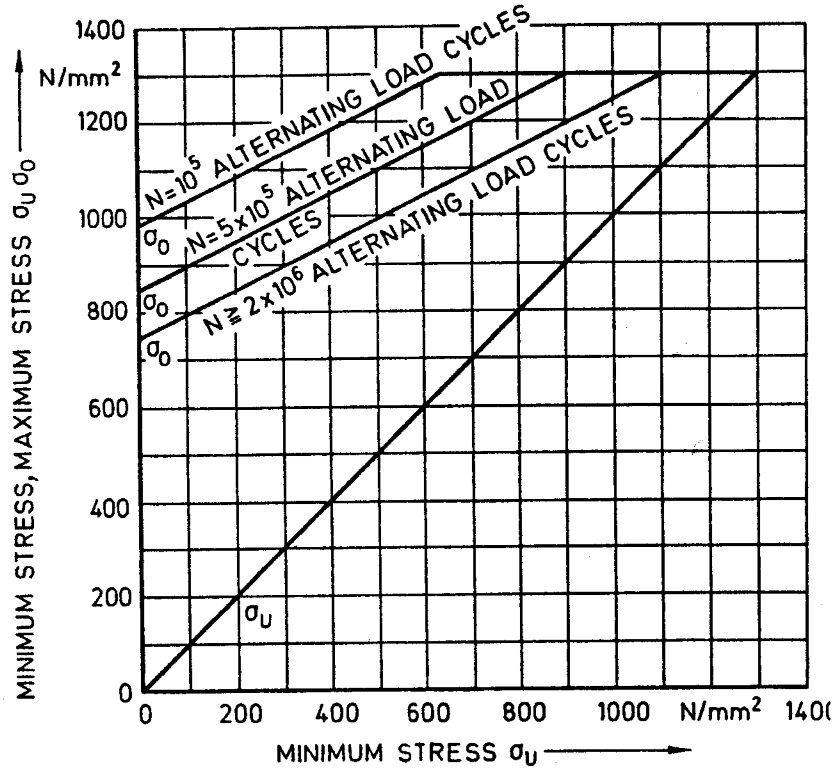


FIG. 13 FATIGUE STRENGTH AND FINITE LIFE FATIGUE STRENGTH DIAGRAM FOR DISC SPRINGS CONFORMING TO IS 12511 (Part 2) (FOR $t < 1$ mm)

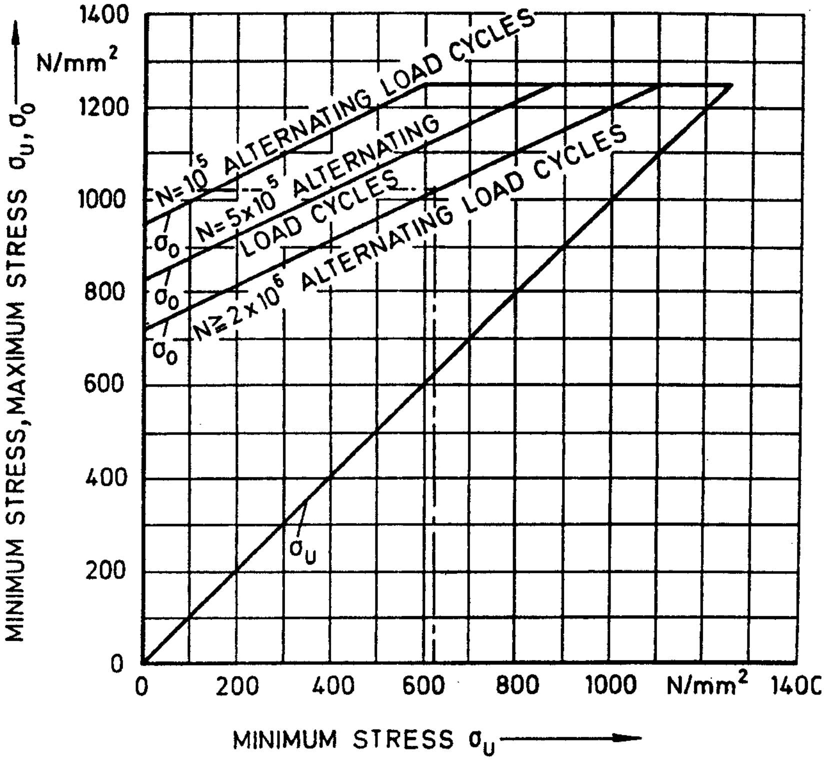


FIG. 14 FATIGUE STRENGTH AND FINITE LIFE FATIGUE STRENGTH DIAGRAM FOR DISC SPRINGS CONFORMING TO IS 12511 (Part 2) (FOR 1 mm ≤ t ≤ 6 mm)

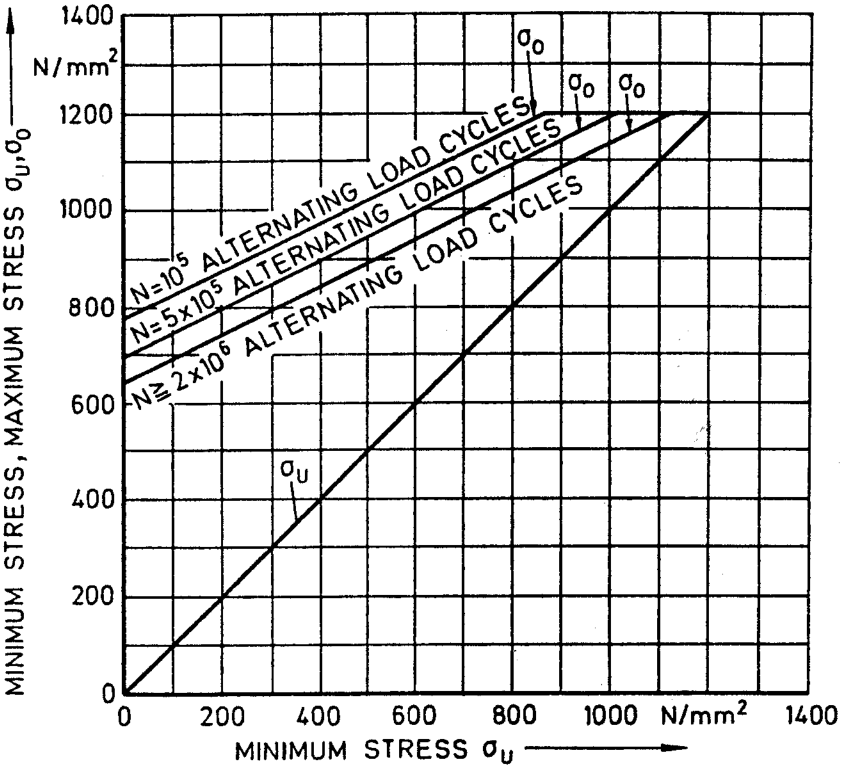


FIG. 15 FATIGUE STRENGTH AND FINITE LIFE FATIGUE STRENGTH DIAGRAM FOR DISC SPRINGS CONFORMING TO IS 12511 (Part 2) (FOR 6 mm ≤ t ≤ 14 mm)

ANNEX A

(*Clause* 9.4)

EXAMPLES OF CALCULATION OF DISC SPRINGS SUBJECTED TO STEADY LOADING OR TO SELDOM ALTERNATING LOADING

A-1 CHECK CALCULATION OF A SINGLE DISC SPRING

A-1.1 In case of disc spring conforming to IS 12511 (Part 2) subjected to steady loading or to seldom alternating loading, it is, as a general rule, not necessary to check the spring force and the stresses at the individual disc by calculation. Such a check calculation can however be carried out with the aid of the equations featured in **5.2**.

A-1.2 The dimensions of disc spring B40GR2 — IS 12511 (Part 2) are for instance (*see* Table 2):

$D_e = 40$ mm, $D_i = 20.4$ mm, $t = 1.5$ mm

$l_o = 2.65$ mm and $h_o = 1.15$ mm

For a spring travel $s_2 = 0.6$ mm,

$$\frac{h_o}{t} = \frac{1.15}{1.5} = 0.77;$$

$$\frac{s_2}{t} = \frac{0.6}{1.5} = 0.4;\ \frac{s_2}{2t} = 0.2$$

$$\delta = \frac{D_e}{D_i} = \frac{40}{20.4} = 1.96 \approx 2$$

From Fig. 4 and Fig. 5

$K_1 = 0.69,\quad K_2 = 1.22,\quad K_3 = 1.38$

A-1.3 Spring force F_2; associated with spring travel s_2, is given by equation (1),

$$F_2 = 905\,495 \times \frac{1.5^4}{1.69 \times 40^2} \times 0.4$$

$$[(0.77 - 0.4)\,(0.77 - 0.2) + 1] = 2\,010 \text{ N}$$

A-1.4 Critical Stress at Location I (*see* Fig. 1)

From equation (2),

$$\sigma_1 = 905\,495 \times \frac{1.5^2}{0.69 \times 40^2} \times 0.4$$

$$[-(1.22\,(0.77 - 0.2) - 1.38] = -1\,530 \text{ N/mm}^2$$

A-1.5 Spring Rate (*see* Table 2)

For $s_2 = 0.6$ mm

From equation (6),

$$R = 905\,495 \times \frac{0.5^3}{0.69 \times 40^2}$$

$$[(0.77^2 - 3 \times 0.77 \times 0.4 + \frac{3}{2} \times 0.4^2 + 1] = 2\,520 \text{ N/mm}$$

A-1.6 Work of elastic strain, for the spring travel from 0 to 0.6 mm. From equation (7),

$$W = \frac{1}{2} \times 905\,495 \times \frac{1.5^5}{0.69 \times 40^2} \times 0.4^2$$

$$[(.77 - 0.2)^2 + 1] \quad = 660 \text{ N/mm}$$

A-1.7 Spring Force in Pressed Flat Condition

From equation (8),

$$F_c = 905\,495 \times \frac{1.5^3}{0.69 \times 40^2} \times 1.15 \ = 3\,180 \text{ N}$$

A-1.8 Spring Travel

The spring travels correlated to various forces can be roughly determined from equation (8) in conjunction with Fig. 6.

Table 2 Dimensions of Disc Spring

(*Clause* A-1.2)

Sl No.	Disc Spring	D_e mm	D_i mm	t mm	h_o mm	l_o mm	F N	s mm	σ N/mm²
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
i)	A 40GR2-IS	40	20.4	2.25	0.9	3.15	6 500	0.68	$\sigma_{II} = 1\,340$
ii)	B 40GR2-IS	40	20.4	1.5	0.15	2.65	2 620	0.86	$\sigma_{III} = 1\,150$
iii)	C 40GR2-IS	40	20.4	1	1.3	2.3	1 020	0.98	$\sigma_{III} = 1\,070$

From Fig. 6,

$$\text{for } \frac{h_0}{t} = 0.77 \approx 0.75$$

the calculation of s_1 relating to F_1 = 1 500 N is as under:

$$\text{With } \frac{F_1}{F_c} = \frac{1\,500}{3\,180} = 0.47 \text{ and } \frac{s_1}{h_0} \approx 0.36,$$

$$s_1 \approx 0.36\ h_0 = 0.36 \times 1.15 = 0.414 \text{ mm.}$$

A-2 CALCULATION OF A SPRING COLUMN

A-2.1 Assume that the spring force of 5 000 N (static loading) is required with spring travel of 10 mm. The diameter of guide pin must not exceed 20 mm.

Hence F_1 = 5 000 N

s_{1ges} = 10 mm

D_i = 20.4 mm [for a 0.4 mm clearance between the disc spring and the guide pin [*see* IS 12511 (Part 2)]

A-2.2 Solution

Starting with the guide pin, a disc spring conforming to IS 12511 (Part 2) is to be selected. Three disc springs are featured for an inside diameter D_i = 20.4 mm, one each of Type A, B and C.

A-2.2.1 The specified conditions (*see* **7.3** and **7.4**) can be met either:

a) With a column of disc spring A 40GR2 — IS 12511 (Part 2) single disc springs if the single disc spring is not exploited to its fullest extent, or

b) With a column of spring assemblies consisting each of two disc springs B 40GR2 — IS 12511 (Part 2).

A-2.2.2 *Calculation with Respect to the Condition Indicated in* **A-2.2.1**(a)

A-2.2.2.1 *Spring force in the pressed flat condition*

From equation (8),

$$F_c = 905\,495 \times \frac{2.25^3}{0.69 \times 40^2} \times 0.9 \quad = 8\,410 \text{ N}$$

A-2.2.2.2 *Spring travel*

The spring travels correlated to various forces can be roughly determined from equation (8) in conjunction with Fig. 6.

With $\frac{F_1}{F_c} = \frac{5\,000}{8\,410} = 0.59$ the spring travel of the single disc relating to the spring force of 5 000 N, $s_1 \approx 0.57 \times h_0 = 0.57 \times 0.9 = 0.51$ mm

A-2.2.2.3 Required number of single discs necessary to achieve the total travel 10 mm.

$$i = \frac{S_{ges}}{s_1} = \frac{10}{0.51} = 19.6$$

20 discs are therefore selected.

A-2.2.2.4 *Dimensions of the spring column*

In this case, the spring column has the following dimensions:

a) Overall height of single disc spring:
l_0 = 3.15 mm;

b) Length of unloaded spring column
$L_0 = i \times l_0 = 20 \times 3.15 = 63$ mm.

c) Length of spring column loaded with
F_1 = 5 000 N
$L_1 = L_0 - s_{1\ ges} = 63{-}20 \times 0.51 = 52.8$ mm.

A-2.2.3 *Calculation with Respect to the Condition Indicated in* **A-2.2.1**(b)

If two single disc springs B 40GR2 — [IS 12511 (Part 2)] are stacked on top of one another to form a spring assembly with n = 2, and a number of these spring assemblies are then piled on top of one another in alternating directions to form a spring column, the requirement can also be met.

A-2.2.3.1 *Spring force for the single disc, ignoring the friction*

$$F_1 = \frac{F_{1ges}}{n} = \frac{5\,000}{2} = 2\,500 \text{ N}$$

A-2.2.3.2 *Spring force in the pressed flat condition*

From equation (8) and **A-1.7**,

F_c = 3 180 N

A-2.2.3.3 *Spring travel*

The spring travels correlated to various forces can be roughly determined from equation (8) in conjunction with Fig. 6,

$$\text{With } \frac{F_1}{F_c} = \frac{2\,500}{3\,180} = 0.79$$

The spring travel of single disc of Type B and of spring assembly amounts to:

$$s_1 \approx 0.71 \times h_0 = 0.71 \times 1.15 = 0.82 \text{ mm.}$$

A-2.2.3.4 The required number of i of spring assemblies necessary to achieve the total spring travel 10 mm.

$$i = \frac{S_{1ges}}{s_1} = \frac{10}{0.82} = 12.2$$

13 spring assemblies are, therefore, selected.

A-2.2.3.5 *Dimensions of the spring column*

In this case, spring column has the following dimensions:

a) Overall height of single disc spring,
$l_o = 2.65$ mm,

b) Overall height of a spring assembly consisting of two disc springs,
$l_o + t = 2.65 + 1.5 = 4.15$ mm,

c) Length of unloaded spring column,
$l_o = i\,(l_o + t)\ 13 \times 4.15 = 54$ mm; and

d) Length of spring column loaded with
$F_{1ges} = 5\,000$ N

$L_1 = L_o - s_{1ges} = 54 - (13 \times 0.82) = 43.34$ mm.

This latter spring column is, therefore, shorter and the individual disc is exploited to greater advantage.

The calculation of stress, spring rate and work of elastic strain is analogous to the example in **A-1**.

Because of the odd number of spring assemblies in this example, one end of the column terminated with D_e, while the other end terminated with D_1. As general rule, an endeavour must be made to let both ends of the column terminate with D_e as illustrated in Fig. 9 and Fig. 10.

ANNEX B

(*Clause* 10.5)

EXAMPLES OF CALCULATION OF DISC SPRING SUBJECTED TO VIBRATORY LOADING

B-1 CHECK CALCULATION OF A SINGLE DISC SPRING

B-1.1 The single disc spring calculated in **A-1** shall be subjected to vibratory stressing between the prestressing force $F_1 = 1\,500$ N and the spring force $F_2 = 2\,010$ N. A check calculation must be made to ascertain whether the disc spring works within the limits of fatigue stroke strength.

B-1.2 In accordance with **A-1**, the values for disc spring B 40GR2 — IS 12511 (Part 2) are as follows:

$F_1 = 1\,500$ N $\qquad s_1 = 0.414$ mm

$F_2 = 2\,010$ N $\qquad s_2 = 0.6$ mm

B-1.3 Theoretical Stress

From Fig. 12, it is observed that in case of disc spring with $\frac{h_o}{t} = 0.766$ and $\delta = 1.96$, the cross-section location σ_{III} is critical in respect of vibratory feature.

a) From equation (4) with spring travel s_1 0.414 mm, the theoretical stress at the underside of the disc spring (location III)

$$\sigma_{III} = 905\,495 \times \frac{1.5^2}{0.69 \times 40^2} \times 0.276 \times 0.51 \times$$

$$[(2 \times 1.38 - 1.22) \times (0.77 - 0.138) + 1.38] =$$

$$610\ \text{N/mm}^2$$

where

$$\frac{s_1}{t} = \frac{0414}{1.5} = 0.276;$$

$$\frac{s_1}{2t} = 0.138;\ \frac{h_o}{t} = 0.77;\ \text{and}$$

$$K_1 = 0.69;\quad K_2 = 1.22;\quad K_3 = 1.38.$$

b) From equation (4)with spring travel $s_2 = 0.6$ mm, the theoretical stress at the underside of disc spring (location III)

$$\sigma_{III} = 905\,495 \times \frac{1.5^2}{0.69 \times 40^2} \times 0.4 \times 0.51 \times$$

$$[(2 \times 1.38 - 1.22) \times (0.77 - 0.2) + 1.38] =$$

$$850\ \text{N/mm}^2$$

$$\frac{s_2}{t} = \frac{0.6}{1.5} = 0.4; \frac{s_2}{2t} = 0.2.$$

B-1.4 Assessment of Fatigue Limit

For the given work travel of disc spring, the values are as follows:

a) Theoretical maximum stress, $\sigma_o = 850$ N/mm^2;

b) Theoretical minimum stress, $\sigma_u = 610$ N/mm^2; and

c) Stroke stress, $\sigma_h = \sigma_o - \sigma_u = 240$ N/mm^2.

From Fig. 14 for $\sigma_u = \sigma_U = 610$ N/mm^2, the value of $\sigma_O = 1\,010$ N/mm^2.

The figure stroke strength amounts to $\sigma_H = \sigma_O - \sigma_U = 400\ \text{N/mm}^2$.

Therefore, $\sigma_h < \sigma_H$ that is the disc spring operates well within its fatigue range.

B-2 CHECK CALCULATION OF A SPRING COLUMN CONSISTING OF SINGLE DISC SPRINGS AND SUBJECTED TO VIBRATORY LOADING

B-2.1 The spring column calculated in **A-2** and consisting of 20 single disc springs A 40GR2 — IS 12511 (Part 2) shall be subjected to vibratory loading between the pressing forces $F_1 = 1\,500$ N and the spring force $F_2 = 5\,000$ N. The spring column shall be verified in respect of fatigue strength.

B-2.1.1 *Spring Force in Pressed Flat Condition*

From equation (8)

$$F_c = 905\,495 \times \frac{2.25^3}{0.69 \times 40^2} \times 0.9 = 8\,410\ \text{N}$$

B-2.1.2 *Spring Travel*

From Fig. 6 corresponding to

$\frac{F_1}{F_c} = \frac{1\,500}{8\,410} = 0.18$ for Types A spring associated with the spring force 1 500 N,

$s_1 \approx 0.155 \times h_0 = 0.155 \times 0.9 = 0.14$ mm,

$s_2 \approx 0.5$ mm (*see* **A-2**).

B-2.1.3 *Theoretical Stress*

From Fig. 12, it is observed that in case of disc spring with $\frac{h_o}{t} = 0.4$ and $\delta = 1.96$, the cross-section location II of the spring is critical in respect of vibratory fracture.

a) From equation (3) with spring travel $s_1 = 0.14$ mm,

$$\sigma_{II} = 905\,495 \times \frac{2.25^2}{0.69 \times 40^2} \times 0.062 \times [-1.22\,(0.4 - 0.031) + 1.38] = 240\ \text{N/mm}^2$$

where

$$\frac{s_1}{t} = \frac{0.14}{2.25} = 0.062,\ \frac{s_1}{2t} = 0.031;$$

$$\frac{h_o}{t} = 0.4 \text{ and } K_1 = 0.69;\ K_2 = 1.22;\ K_3 = 1.38.$$

b) From equation (3) with spring travel, $s_2 = 0.5$ mm,

$$\sigma_{II} = 905\,495 \times \frac{2.25^2}{0.69 \times 40^2} \times 0.22 \times [-1.22\,(0.4 - 0.11) + 1.38] = 937\ \text{N/mm}^2$$

where

$$\frac{s_2}{t} = \frac{0.5}{2.25} = 0.22,\ \frac{s_2}{2t} = 0.11$$

B-2.1.4 *Analysis of Fatigue Limit*

$\sigma_h = \sigma_o - \sigma_u = 937 - 240 = 697\ \text{N/mm}^2$ when the value of σ_u corresponding to $\sigma_o = 240\ \text{N/mm}^2$ is determined from Fig. 14, it is observed $\sigma_h > \sigma_H$.

The spring is situated within the region of finite life fatigue strength of 5×10^5 alternating load cycles.

In the case of a spring column consisting of 20 single disc springs, the strength shown in the diagram are not quite attained (*see* **10.4.1**). A slightly shorter service life must, therefore, be anticipated.

If the disc springs are to retain full fatigue strength and the maximum stress $\sigma_o = 937\ \text{N/mm}^2$ is to be maintained then according to Fig. 14, allowing the safety margin, the minimum stress amounts to

$$\sigma_u \approx 500\ \text{N/mm}^2$$

By interpolation, the required prestressing spring travel becomes

$$s_1 > \frac{500}{240} \times 0.14 = 0.29\ \text{mm}$$

B-2.1.5 *Necessary Correlated Spring Force*

$$F_1 \approx 0.35 \times F_c = 0.35 \times 8\,410 = 2\,940\ \text{N}.$$

where

$F_c = 8\,410$ N (*see* **B-2.1.2**)

$\frac{F_1}{F_c} \approx 0.35$ (from Fig. 6 corresponding to

$$\frac{s_1}{h_o} = \frac{0.29}{0.9} = 0.32$$

B-2.1.6 *Permissible Maximum Stress, Permissible Spring Travel and Correlated Spring Force*

a) If the minimum stress $\sigma_u = 240\ \text{N/mm}^2$ is retained, the permissible maximum stress allowing for a safety margin can be read off from Fig. 14.

$$\sigma_o \approx 800\ \text{N/mm}^2$$

b) Permissible spring travel can be obtained by interpolation

$$s_2 < \frac{800}{937} \times 0.5 = 0.43\ \text{mm}$$

c) The correlated spring force can also be calculated with the help of Fig. 6.

$$\frac{s_2}{h_o} = \frac{0.43}{0.9} = 0.48$$

$$F_2 = 0.51 \times F_c = 0.51 \times 8\,410 = 4\,290 \text{ N.}$$

B-3 CHECK CALCULATION OF SPRING COLUMN CONSISTING OF SPRING ASSEMBLIES AND SUBJECTED TO VIBRATORY LOADING

B-3.1 The spring column calculated in **A-2** and consisting of 13 assemblies of two disc springs each, B 40GR2 — IS 12511 (Part 2) shall be subjected to vibratory loading between the prestressing force F_1 = 1 500 N and the spring force F_2 =5 000 N. The spring column shall be verified in respect of fatigue strength.

B-3.1.1 *Spring Force for the Single Disc Ignoring the Friction*

$$F_1 = \frac{F_{1\text{ges}}}{n} = \frac{1\,500}{2} = 750 \text{ N}$$

B-3.1.2 *Spring Force in the Pressed Flat Condition*

From equation (8) and **A-1.7**

$$F_c = 3\,180 \text{ N.}$$

B.3.1.3 *Spring Travel*

The spring travels correlated to various forces can be roughly determined from equation (8) in conjunction (*see* Fig. 6) with

$$\frac{F_1}{F_c} = \frac{750}{3\,180} = 0.24$$

Spring travel of single disc spring for Type B

$$s_1 \approx 0.17 \times h_o = 0.17 \times 1.15 = 0.20 \text{ mm,}$$

$$s_2 \approx 0.78 \text{ (}\textit{see}\text{ \textbf{A-2})}$$

B-3.1.4 *Theoretical Tensile Stress*

In accordance with Fig. 12, cross-section III is the determining one for vibratory fracture.

From equation (4)

$$\sigma_u = 308 \text{ N/mm}^2 \text{ when } s_1 = 0.20 \text{ mm, and}$$

$$\sigma_o = 1\,060 \text{ N/mm}^2 \text{ when } s_2 = 0.78 \text{ mm.}$$

The service life of the disc spring will be in the region of 10^5 alternating load cycles approximately.

B-3.1.5 *Spring Characteristic*

Because the spring column, consists of spring assemblies each comprising two stacked spring, that is, 26 individual disc springs in all, the values known in the fatigue strength diagrams can only be used if adequate safety margins are adopted (*see* **10.4.1**).

The spring column consisting of single disc spring A 40GR2 — IS 12511 (Part 2) is superior to the spring column consisting of two stacked spring B 40GR2 — IS 12511 (Part 2) in respect of length of service life when subjected to vibratory loading.

ANNEX C

(Foreword)

COMMITTEE COMPOSITION

Automotive Springs and Suspension Systems Sectional Committee, TED 21

Organization	*Representative(s)*
Tata Motors Ltd, Jamshedpur	Shri A. G. Pradhan ***(Chairman)*** Shri K. Gopalakrishna *(Alternate)*
Akal Springs Pvt Ltd, Ludhiana	General Manager
All India Springs Manufacturers Association, Mumbai	Shri A. A. Mirchandani
Ashok Leyland Ltd, Chennai	Shri Appalaraju Shri U. Jaikrishna *(Alternate)*
Association of State Road Transport Undertakings, New Delhi	Shri A. S. Lakra Shri P. M. Phate *(Alternate)*
Central Institute of Road Transport, Pune	Shri N. R. Kachare Shri P. S. Munoli *(Alternate)*
Central Mechanical Engineering Research Institute, Durgapur	Dr J. Basu Dr T. K. Paul *(Alternate)*
Conventry Springs & Engineering Co Pvt Ltd, Kolkata	Shri A. Bafna Shri A. S. Kohli *(Alternate)*
Controllerate of Quality Assurance (CQA) (OFV) Vehicle Factory, Jabalpur	General Manager
Gabriel India Ltd, Mumbai	Shri K. Sundararaman Shri S. K. Bhaumick *(Alternate)*
Jai Parabolic Springs Ltd, Chandigarh	Shri Sunil Haroliya
Jamna Auto Industries Ltd, Yamuna Nagar	Shri D. S. Gill Shri B. K. Khandelwal *(Alternate)*
Kemen Springs Pvt Ltd, Mumbai	Shri P. K. Mirchandani
Mack Springs Pvt Ltd, Thane	Shri D. V. Sharma
Mahindra & Mahindra Ltd, Nashik	Shri Ravindra Deshmukh Shri Kailash Jat *(Alternate)*
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Ministry of Heavy Industry & Public Enterprises, New Delhi	Shri S. K. Bharij Shri R. K. Tripathi *(Alternate)*
Research Designs & Standards Organization, Lucknow	Joint Director (Standards) Assistant Design Engineer
Stumpp, Schuele & Somappa Pvt Ltd, Bangalore	Shri B. S. Mookherjee Shri N. C. Srinivasan *(Alternate)*
The Automotive Research Association of India, Pune	Shri S. Raju
Upper India Steel Manufacturing & Engineering Co Ltd, Ludhiana	Shri R. P. Engira
Vehicle Factory, Jabalpur	Shri M. K. Mishra Shri R. Lodwal *(Alternate)*
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